CLARK & WESTBAY

Firebox Design and its Relation to Boiler Performance

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FIREBOX DESIGN AND ITS RELATION TO BOILER PERFORMANCE

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THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

CHARLES M. CLARK AND JAMES HERRON WESTBAY

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FIREBOX DESIGN AND ITS RELATION TO BOILER PERFORMANCE.

Contents.

FIREBOX DESIGN AND ITS RELATION TO BOILER PERFORMANCE.

Introduction.

The performance of a locomotive boiler is dependent to a very large extent upon the design of its firebox. A great deal of experimental work has been done in an effort to determine what steps should be taken to improve boiler performance and to increase boiler efficiency. As a result of this experimental work, various improvements have been made in locomotive boilers, but they have been confined largely to the boiler proper, as distinguished from the firebox.

Various test data have revealed the fact that under average conditions the firebox of a locomotive is forced to perform at a point very near to its maximum capacity, while at the same time the boiler, or tube heating surface, is working far below its capacity. The conclusion that has been drawn, therefore, is that the maximum rate at which a locomotive boiler can be worked is limited by the ability of the firebox to furnish heat to it; in othor words, the capacity of the firebox limits the performance of the boiler. It would seem, then, that there is in firebox design a fruitful field for investigation and improvement, so that ultimate boiler capacity may be increased. It is with such an end in view that an attempt is here made to investigate the relation which firebox design bears to boiler performance, and to suggest some means by which firebox design may be improved. Various test data have been analyzed, bulletins and articles bearing on the subject have been reviewed and standard texts have been consulted in an effort to determine where these

improvements in locomotive fireboxes can be most effectively made.

Purpose.

This paper is written in an effort to investigate the losses which take place in a locomotive furnace and, if possible, to discover some means by which the overall efficiency of a locomotive boiler may be increased by improvements in firebox design.

PART I

Theory of Combustion

Definition.

Combustion is a rapid process of oxidation. It takes place when any gas, liquid, or solid combines with oxygen at a high temperature.

The Combustion Process and Heat Distribution.

In the following discussion, the fuel dealt with is coal, although oil is in some localities used for locomotive fuel. Coal is made up of a number of constituents, the analysis of which may be classified in two ways, viz., proximate analysis, and ultimate analysis. In the proximate analysis of coal, the percentages of the following constituents are determined: fixed carbon, volatile matter, moisture, and ash. The ultimate analysis consists in the determination of the percentages of the following elements, all of which are present, in varying amounts: nitrogen, oxygen, hydrogen, carbon, sulphur, ash, moisture. An understanding of the constituents of coal as indicated by the preceding forms of analyses is of valuable assistance in studying the theory of combustion. A detailed analysis of the effect of each constituent upon combustion will be considered later.

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When coal is placed in a furnace it undergoes certain physical and chemical changes as combustion proceeds. No definite information is available regarding the physical changes which take place in the coal after it is fired into the furnace. It is known, however, that these changes vary with different grades of coal and designs of firebox. The fixed carbon in the coal remains upon the grate and burns with a short flame, while the volatile matter distils off in the form of gas and burns with a long flame. The variation in draft due to intermittent exhausting of the engines may have a marked effect upon the fire. An intense draft may actually lift the fire from the grates, and thus allow cold air to enter the firebox through the unobstructed openings in the grate. An intense draft will cause a high cinder loss, because it will draw small particles of unburned coal out through the tubes, and send them out from the stack.

After the coal has been placed upon the grates certain definite chemical changes take place. Unlike the physical changes, the chemical changes are more or less definitely known, and the laws governing them may be discussed. Average grades of bituminous coal contain from 6% to 12% moisture. Immediately after reaching the grates the temperature of the coal is raised, and the moisture is driven off in the form of steam. This moisture in being changed to steam, absorbs heat from the furnace, the amount of heat so absorbed being determined by the pressure in the firebox and by the temperature of the furnace gases. Under any circumstances this steam becomes highly superheated, since the pressure in the locomotive furnace is slightly below that of the atmosphere. This superheated steam mixes with the

furnace gases and goes with them through the tubes, giving up some of its heat to the boiler. It passes out through the stack as superheated steam at the temperature of the smokebox. Hence, as superheated steam, it contains the latent heat of evaporation, and represents a complete loss. The amount of heat thus lost per pound of coal as fired, may be determined by use of the following equation:

$$C_1 = \left[q + r + C_p(T-t) - h\right] x \frac{\%\text{ moisture in coal}}{100}, \text{ where} \qquad (1)$$

C_1=loss in B. T. U. per lb. of coal as fired,

q = heat of the liquid at smokebox pressure,

r = latent heat of evaporation at smokebox pressure,

C_p= specific heat of superheated steam at constant pressure,

T = smokebox temperature, degrees F,

t = temperature of steam at smokebox pressure, degrees F,

h = heat in the liquid at atmospheric temperature.

As noted in the ultimate analysis, coal contains a small amount of hydrogen, the amount varying to some extent with the percentage of volatile matter present in the coal. As soon as the temperature of the coal is sufficiently high the volatile matter distils off in the form of gases. During this process free hydrogen is liberated from the coal and at once combines with eight times its weight of oxygen, thus forming water. Approximately 62,000 B. t. u. are liberated when one pound of hydrogen combines with eight pounds of oxygen, forming nine pounds of water. All of the heat thus liberated is not available for absorption by the boiler, for the water which results from this combination of hydrogen and oxygen passes out through the stack as superheated steam. The heat contained in this nine

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pounds of steam must be subtracted from 62,000 B.t.u. in order to find the amount of heat which is made available by this chemical combination. The heat contained in the superheated steam may be found by use of equation (1), in which the percentage factor must be adjusted. The hydrogen content of coal is usually low and therefore the amount of heat realized from this combination of hydrogen and oxygen is not large. The exact amount of heat thus made available may be determined by the following equation:

$$C_2 = [62,000-9(S-h)] \times \frac{\%\text{ of hydrogen in coal}}{100}, \text{ in which} \qquad (2)$$

C_2 = available heat per lb. of coal, due to burning of hydrogen,

S = total heat in escaping steam,

h = heat of the liquid at atmospheric temperature,

The air necessary for combustion is taken from the atmosphere surrounding the firebox. This air always contains moisture, which in reality is moist steam at atmospheric temperature. After this moist steam enters the firebox it is superheated, in which state it remains until exhausted from the stack. The heat contained in this superheated steam is completely lost, since it was taken from the firebox and not given up to the boiler. This heat loss is determined as follows:

$$C_3 = [S-(q+r)] \times \frac{\%\text{ of moisture in air}}{100}, \text{ where} \qquad (3)$$

C_3 = heat taken from furnace, per lb. of air entering firebox,

S = total heat in superheated steam,

q = heat of the liquid at atmospheric temperature,

r = latent heat of evaporation at atmospheric temperature.

The volatile gases, which consist for the most part of oxygen, carbon, and hydrogen, distill off at a comparatively low temperature. These hydrocarbons burn with a long flame and in order to produce efficient combustion, they must be mixed thoroughly with a sufficient supply of oxygen. At present comparatively little is known of the exact composition of these hydrocarbons, or of their heating value. The amount of hydrocarbons contained in coal varies with the kind of coal. Coals high in fixed carbon, for example, distill off a comparatively small amount of volatile gases, while coals which are mined in Illinois, or similar bituminous coals which have a low fixed carbon content are higher in volatile matter. Little attention has been given to the proper methods of burning these volatile gases in locomotive fireboxes. Since these gases burn with a long flame, it is obvious that a firebox of large volume must be used, and because a thorough mixing of these gases with oxygen is necessary, adequate means must be provided to complete this process.

After the volatile gases pass out of the flues into the smokebox they are exhausted into the atmosphere, although they still contain a large amount of heat which might have been absorbed by the boiler. This loss of heat is known as the stack loss, and by means of equation (4) given below, its magnitude may be determined.

$$H_1 = C_p(T-t) \tag{4}$$

H_1 = heat, above atmospheric temperature, in 1 lb. of gas.

C_p = specific heat of the gas at constant pressure,

T = smokebox temperature, degrees F,

t = atmospheric temperature, degrees F.

It must be remembered that the above formula gives the amount of heat contained in the gases, above atmospheric temperature. Although this is a total loss, the boiler could not possibly have absorbed all the heat contained in the exhaust gases. The following equation, however, represents that part of the heat in the exhaust gases which might have been absorbed by the boiler:

$$H_2 = C_p(T - t_1) \qquad (5)$$

H_2 = heat contained in 1 lb. of gas, above boiler water temperature.

C_p = specific heat of the gas at constant pressure.

T = smokebox temperature, degrees F.

t_1 = boiler water temperature, degrees F.

The difference between H_1 and H_2 represents a loss which can under no circumstances be eliminated. A graphical representation of this loss is shown in Fig. 3, page 24.

The fixed carbon in coal burns upon the grate at a much higher temperature than that at which the volatile gases distil. The laws which govern the burning of carbon are well known and the heat resulting from such combustion may be accurately determined. The chemical equation given below represents the transformation completed in the firebox, when an adequate supply of air is available:

$$\underset{12}{C} + \underset{32}{O_2} = \underset{44}{CO_2}. \qquad (6)$$

In other words, for every pound of carbon burned, 2-2/3 lbs. of oxygen are necessary to complete the process. When 1 lb. of

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carbon is so burned, 14,600 B.t.u. are liberated. If carbon is burned in an insufficient supply of oxygen, carbon monoxide is formed, as shown by equation (7):

$$\underset{12}{C}+\underset{16}{O}=\underset{28}{CO}. \tag{7}$$

When 1 lb. of carbon is thus burned to carbon monoxide, 4,451 B.t.u. are liberated. Therefore 10,149 B.t.u. are lost by burning 1 lb. of carbon in an insufficient supply of oxygen. This shows that it is imperative to have at all times a sufficient supply of oxygen to complete the burning of the carbon, as well as the volatile matter contained in the coal.

In all commercial furnaces air furnishes the supply of oxygen. For every pound of oxygen that is drawn into the firebox 3.32 lbs. of nitrogen must necessarily accompany it. This nitrogen, being an inert gas, has little or no effect upon the combustion which takes place in the firebox. It has, however, a very marked effect upon the efficiency of the furnace. It is drawn into the firebox at atmospheric temperature, is heated to firebox temperature and then passes out with the volatile gases through the tubes, and is exhausted from the stack at smokebox temperature. Like the moisture loss, this heating of the nitrogen represents heat taken from the firebox, and it constitutes a part of the stack loss. Equation (8) expresses this loss due to exhausting the heated nitrogen:

$$H_3=C_p(T-t) \tag{8}$$

H_3=heat contained in 1 lb. of nitrogen as exhausted,

C_p=specific heat of nitrogen at constant pressure,

T =temperature in smokebox, degrees F,

t =temperature of atmosphere, degrees F.

This nitrogen loss represents one of the greatest losses which take place during the process of combustion. As heretofore stated, 2-2/3 lbs. of oxygen are necessary to burn 1 lb. of carbon to carbon dioxide, and this oxygen is accompanied by 8.85 lbs. of nitrogen, which must be heated and passed out through the stack. In burning 1 lb. of hydrogen to water 26.5 lbs of nitrogen must be so heated. On account of the relatively small amount of free hydrogen in the volatile gases the latter amount of nitrogen is of no great importance in average locomotive practice.

As previously stated, a deficiency in the air supply will produce an abnormal amount of CO, or carbon monoxide. For every pound of carbon burned to CO, there is a heat loss of about 10,150 B.t.u. If the supply of air is greatly in excess of the amount required the firebox temperature will be lowered, and the overall boiler efficiency will be decreased. In any combustion process which takes place in a furnace the actual temperature maintained is far below the possible theoretical temperature. This is due to radiation, incomplete combustion, time limitations, stack losses, etc. The theoretical temperature available may be found by the following formula:

$$\text{Increase in temperature} = \frac{\text{B.t.u. generated by combustion}}{\text{Wt. of products of combustion} \times C} \quad (9)$$

If a firebox is so designed that an undue amount of CO is formed the temperature will be low, the same as if an excess of oxygen were present. In actual practice there must always be an excess supply of air in the firebox, in order to insure good combustion. The amount desired in excess of the theoretical amount varies under different conditions. Tests have shown that

an excess supply of from 15 to 25% of air is not harmful. If a volumetric analysis of flue gas be obtained the dry air used per pound of carbon consumed could be calculated from the following formula:

$$W = \frac{3.032N}{CO_2 + CO}, \text{ where} \qquad (10)$$

N, CO_2, and CO are percentages by volume,

W = Wt. of dry air in lbs. per lb. of carbon burned.

If equation (10) be multiplied by the percent of fixed carbon in the coal the weight of air used per pound of dry coal will be found. The amount of air used, as determined from equation (10) will always be in excess of the theoretical amount required if all the carbon is to be burned to CO_2. Ideal conditions would be such that just enough oxygen be admitted to burn all the carbon to CO_2, all the hydrogen to H_2O, and to take care of the volatile gases. It is impossible to do this, although the excess supply should be as small as possible. Since the amount of air which can be drawn into the firebox is limited by the ash pan openings and the draft, these features should receive careful attention in order that the openings should be of the proper size. The imperfect combustion in many locomotives may be due to the fact that not enough air can be drawn through the ash pan openings to supply the fire. The style of grates used has a marked effect upon the air supply. The ratio of area of air-openings to total grate area should be as large as possible.

In the foregoing analysis of the losses which take place during the process of combustion no mention was made of ashpan losses, radiation and "unaccounted for." Even the most carefully designed grates allow coal to drop through the openings, and to

fall down into the ashpan. Usually this is a loss which is never recovered. Since the temperature of the locomotive firebox is much higher than that of the surrounding atmosphere there must be a loss due to radiation of heat through grates and firedoor. The parts of the firebox which are surrounded by water absorb all of the heat radiated to their surfaces.

Summary.

In summing up the more important points concerning combustion in locomotive fireboxes the following should be emphasized:

(1) Proper combustion is impossible unless a sufficient supply of air is available at all times. If the air supply is too great the firebox temperature will be too low. If the air supply is too small incomplete combustion, with formation of CO takes place, and the temperature is again too low.

(2) In addition to providing a sufficient air supply the firebox design should be such that the air may become thoroughly mixed with other products of combustion.

(3) The longer the gases can be kept in the firebox the more complete will be the combustion. This is due to the fact that the time element is of great importance in relation to efficiency of combustion.

In applying these principles and the facts brought out by the study of combustion in the locomotive furnace, the following items should receive particular attention in firebox design:

(1) The ratio of air openings in the grate to grate area should be as large as possible.

(2) The ashpan should be so designed that a sufficient

supply of air can be obtained at all times.

(3) The path of the gases, and consequently the time they remain in the furnace, should be lengthened. This can be done by use of the firebrick arch and the combustion chamber. (Firebrick arches aid in maintaining furnace temperatures and provide effective radiating surfaces. Combustion chambers furnish valuable heating surface in addition to the functions just mentioned. These matters are considered elsewhere in this paper.)

(4) The ratio of firebox volume to firebox heating surface should be as large as possible. A very satisfactory method of realizing this condition is by the installation of a combustion chamber.

(5) The ratio of firebox volume to grate area should be as large as pssible.

If the above items were carefully observed at the time the firebox was designed, doubtless more complete combustion could be realized. It would be possible to burn coal at a higher rate, and do so efficiently. It must be remembered that the firebox and not the boiler limits the capacity of the locomotive. Therefore, any increase in the efficiency of combustion will act directly to increase the capacity of the locomotive.

PART II.

Heat Transmission in Locomotive Fireboxes.

Heat is transferred from the fire to the boiler water by three different methods, viz., Radiation, Conduction, and Convection.

Radiation.

Radiation may be defined as the transfer of heat from one body to another one some distance away, without the aid of an intervening medium. The transfer is made by heat waves, and it takes place directly between the two bodies. The amount of heat transferred by radiation does not depend upon the intervening medium, but upon the radiating surface, the absorbing surface, and the form of these two surfaces. The heat transfer between the bodies per unit of time depends, to some extent, on the distance between them. The amount of heat transferred by radiation per unit of time, for a unit area of radiating surface, is given by the following formula:

$$H = C(T^4 - T_1^4), \text{ where} \tag{1}$$

H = heat transfer (in B.t.u.) per unit area per unit of time,

T = furnace temperature, degrees F, absolute,

T_1 = boiler plate temperature, degrees F, absolute,

C = a constant.

The constant, C, varies with different furnaces. It is constant for each firebox, and depends upon the characters of the radiating and the heat-absorbing surfaces, and the distance between them. Since this constant is determined for a certain furnace, the amount of heat which it transfers in a unit of time depends merely upon the temperatures of the furnace and the heat-absorbing plates. Since the amount of heat transferred varies as the difference of the fourth powers of these temperatures, it is desirable that a high firebox temperature be maintained. Radiation is of great importance when discussing the relative evaporative values of different parts of the heating surfaces. The greater

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part of the heat which goes to the firebox heating surface is transferred by radiation. Since almost one-fourth of the total steam generated in a locomotive boiler is generated by the firebox heating surface, it is obvious that any feature of firebox design that will promote heat transfer by means of radiation is of importance. The firebox heating surface is the only part of the heating surface that receives any great amount of heat by radiation.

Conduction.

Conduction may be defined as the transfer of heat from a hot body to a cooler one by a physical contact between the two bodies. All of the heat that is transferred from the fire to the boiler water must pass through either the firebox sheets or the boiler tubes. This heat is transferred through the firebox sheets, the tubes and flues by means of conduction. The amount of heat thus transferred per unit area of heating surface, in a unit of time, may be expressed as follows:

$$H = \frac{C}{d}(t_1 - t_2), \text{ where} \qquad (2)$$

H = heat, in B.t.u., transferred by conduction, per unit area, in a unit of time.

t_1 = temperature of dry surface of plate, degrees F.

t_2 = temperature of wet surface of plate, degrees F.

d = distance between wet and dry surfaces of plate.

C = a constant.

The constant, C, depends upon the conductivity of the metal of which the plate is made. If a plate is covered with soot or scale, as most boiler plates and tubes are, this fact should be taken into account when determining the values of C and d. It has been found by experiment that the difference in temperature

between the dry and wet sides of a plate does not have to be great in order to transfer a large amount of heat. The presence of soot and scale on the plates probably does not interfere to any very great extent with the transfer of heat. It follows, then, that the greater part of the temperature loss occurs just before the heat reaches the plate. Fig. 1, page 16, illustrates the temperature gradient in a boiler plate, and shows that there is only a small difference in temperature of the two sides of the plate.

Convection.

Convection is defined as the transfer of heat from a hot body to a cooler one by some medium, usually a gas. In a locomotive furnace the gases absorb their heat from the firebox, and then pass through the tubes and flues. As they pass over the heating surfaces the hot particles of gas strike the boiler plates and impart heat to them. Convection is a process of continuous interchange of position between cooled particles of gas next to the heat-absorbing surface and the hotter particles within the body of the gas. Nearly all of the heat that is transferred through the tubes and flues is by means of convection, since it is over these surfaces that the hot gases pass.

Equation (3) may be used to determine the amount of heat transferred by convection for a unit area of heating surface in a unit of time:

$$H = A(T-t_1)+Bav(T-t_1), \text{ where} \qquad (3)$$

A and B are constants depending upon the medium, etc.

T = temperature of gases in the firebox, degrees F.

t_1 = temperature of dry side of boiler plate or tube, degrees F.

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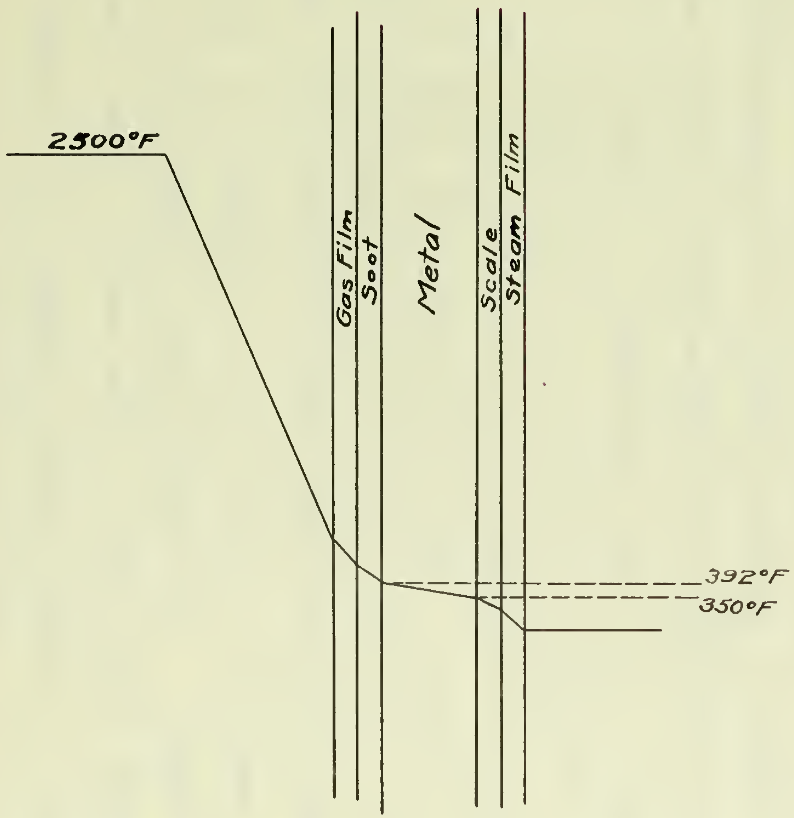
Temperature Gradient in Firebox and Boiler.



Paths of Heat Transfer in Firebox and Boiler.

Plates Free from Soot or Scale.

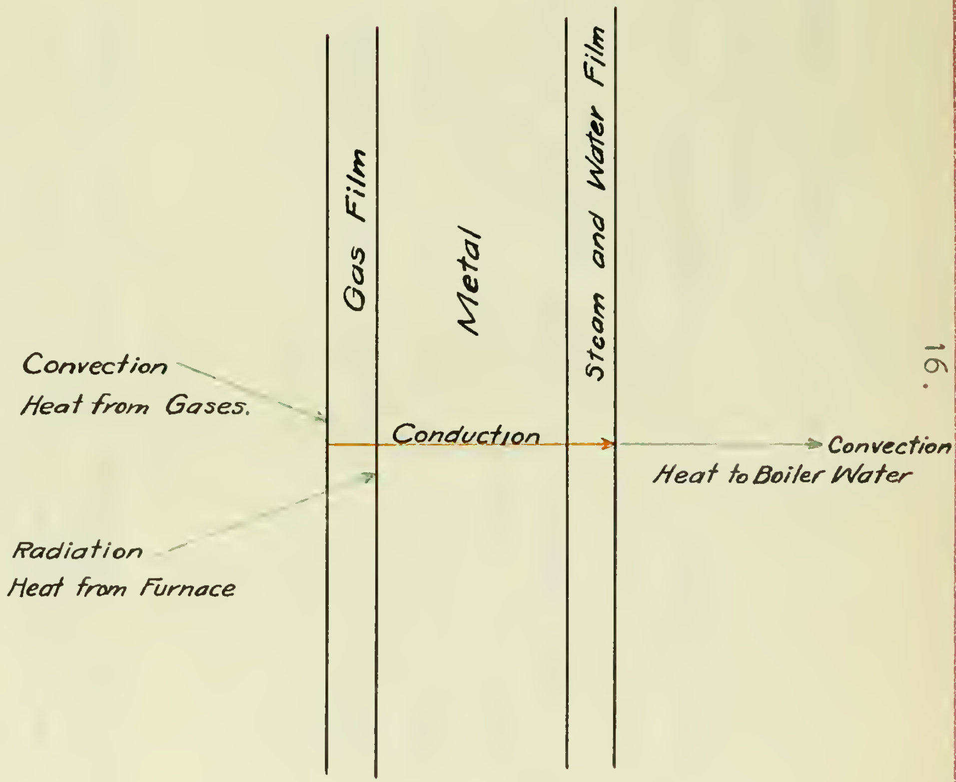


FIG. 1. – HEAT TRANSFER.

a = density of medium, or gas.

v = velocity of gas.

The first part of the equation, $A(T-t_1)$ takes care of the temperature difference. This part of the equation is the more important when dealing with boilers in which the gases have a very low velocity. In locomotive boilers, however, the gases attain a very high velocity, so high, in fact, that the velocity factor, $Bav(T-t_1)$, becomes far more important than the factor $A(T-t_1)$. Therefore, the latter factor may be ignored, and the equation written:

$$H = Cav(T-t_1), \tag{3a}$$

for use in dealing with locomotive boilers. From equation (3a) it is clear that the amount of heat transferred per unit of heating surface in a unit of time is proportional to velocity of the gases in passing over this heating surface. When the velocity of the gases exceeds the critical velocity it seems that there is a higher relation between the heat transferred and the first power of the velocity. Experiments show that the heat transfer increases as some power of the velocity, this power being between 1 and 2. The critical velocity of a gas may be defined as that velocity at which the gas tends to leave a straight-line motion and begins to spread and break up into eddies and whirls. For most gases this velocity is about 2000 feet per second. As might be expected, an increase in the velocity of the gases causes an increase in the amount of heat transferred. A higher velocity of the gases tends to sweep away an inert layer of gas which usually clings to the heating surface, forming a very effective insulator from heat. Referring to equation (3), page 15, it

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is seen that the amount of heat transferred is directly proportional to the density of the hot gases. This follows directly from the fact that the greater the number of particles in a unit volume of gas, the greater will be the number of particles that will strike the heating surface in a unit of time. The amount of heat transferred depends also upon the temperature difference, $(T-t_1)$.

The velocity factor has not been properly considered in the designing of many locomotive boilers. If the tubes in a boiler are made smaller than those in common use at present the heat transfer from the gases to the boiler water will be affected in the following ways:

(1) The velocity of the gases passing through the tubes will be increased, if equal weights of gas must pass through in a unit of time. This will, as can be seen, increase the rate of heat transfer.

(2) As a result of the gases going through the tubes at a high velocity, the layer of inert gas usually found near the surface of the tubes will be swept away, thus allowing a closer contact of the hot gases with the tube surfaces.

(3) The hot gases in the middle of the tube will be more likely to strike the tube walls and give up their heat. Experiments show that heat transfer is inversely proportional to the radius of the tube. This would indicate, therefore, that a tube of infinitesimal diameter would transfer an infinite amount of heat. There are practical considerations, however, which limit the minimum diameter of tubes which can be used in locomotive boilers. High maintenance costs, excessive leaking, and danger

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of choking by cinders prohibit the use of tubes less than about 1-1/2 inches in diameter. Experiments conducted by Mr. Henry Kreisinger, of the Bureau of Mines, show that if two tubes of the same length but one having twice the diameter of the other, conduct equal amounts of gas in the same time, the smaller tube transfers the greater amount of heat. In some instances the smaller tube transferred twice as much heat as the larger one.

Within certain limits the heat transfer in a tube is directly proportional to the length of the tube. But here, again, are practical considerations which fix the maximum length of tubes for locomotive boilers. Standard practice limits this length to about 21 feet, although a few locomotive boilers having 22 foot tubes are in service. Tubes of this length sag dangerously. It is doubtful if such long tubes are advisable, because they allow the gases to cool too much before reaching the smokebox. The temperature of the gases in the smokebox should be at least as high as that of the superheated steam, and preferably higher.

Graphical Representation of Heat Transfer.

The first diagram of Fig. 1, page 16, shows the temperature gradient as the heat passes from the firebox to the boiler water. As shown by this figure, a very large temperature drop occurs between the firebox and boiler water. The greater part of this temperature drop takes place at the surface of the plate, and the presence of soot and scale does not seem to cause much additional decrease of temperature. The temperature drop between the two surfaces of heating plate is very small, usually being less than 50° F.

The second diagram of Fig. 1, page 16, shows how the heat is

transmitted from the firebox to the boiler water. The plate shown is free from soot and scale. Heat is radiated directly from the firebox to the surface of the boiler plate, and the heat from the gases is transferred by convection to the layer of inert gas which lies along the surface of the plate, as shown in the diagram. All of the heat on the dry surface of the plate is transferred by conduction through the layer of inert gas, the boiler plate, and the layer of steam bubbles on the wet surface of the plate. Finally, the heat is transmitted to the boiler water by convection.

Summary.

The following points should be emphasized in considering heat transmission:

(1) The quantity of heat which is transmitted to the boiler tubes by radiation is small.

(2) The firebox heating surface receives nearly all of its heat by radiation.

(3) By far the greater part of the heat which is received by the tubes is transmitted by convection.

(4) The boiler plates or tubes are able to transmit any amount of heat which they receive. The presence of scale or soot upon plates or tubes prevents a certain amount of heat from reaching the metal and being transmitted by it. The exact amount of heat losses due to soot and scale are not well known.

(5) The difference in temperatures of the dry and the wet surfaces of a plate need not be great, about 40° F. being sufficient.

(6) Unsatisfactory conditions as to heat transfer by con-

vection are often responsible for low boiler efficiency. Increasing the velocity and density of the gases will increase heat transfer by convection.

PART III.

Heat Distribution and Furnace Efficiency.

Discussion of Articles by Mr. Anthony.

The material used in the sections on combustion and heat transmission was in part derived from technical journals and standard texts. Of the articles so reviewed those written by Mr. J. T. Anthony on "Locomotive Boiler Efficiency" have proven so helpful that special attention will be given them here. A number of diagrams have also been reproduced from Mr. Anthony's papers.

Figure 2, page 22 shows heat balances at different rates of combustion. The curves were derived from tests of four different locomotives, all of modern design. Their boilers were equipped with firebrick arches and superheaters, and are representative of modern improved locomotive boilers. The fuel used during these tests was a high grade of bituminous coal in which the volatile content was about 35% and the heating value was about 14,000 B.t.u. per pound. The losses at various rates of combustion are well illustrated by the chart referred to. The loss due to radiation increases uniformly with the rate of combustion, increasing from about 1-1/2% to 10% as the combustion rate increases from 30 to 180 lbs of coal per square foot of grate per hour. The loss due to combustible in ash remains constant at about 1-1/2% for all rates of combustion. The spark and

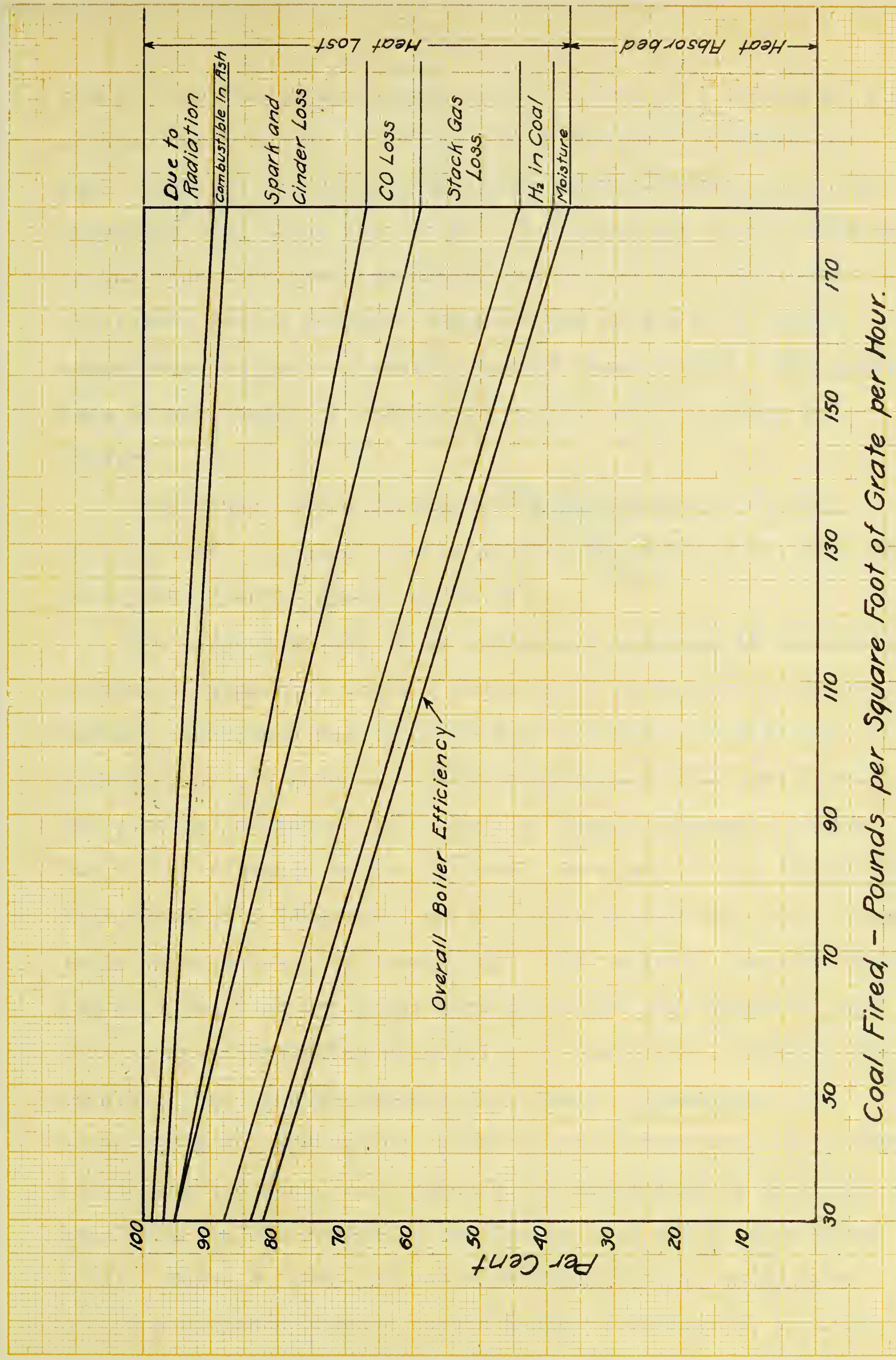

FIG. 2. - EFFECT ON LOSSES AND EFFICIENCY OF INCREASING COMBUSTION RATE.

cinder loss increases rapidly as the firing rate increases, and at high rates amounts to about 20% of the total heat in the coal. Between the combustion limits shown the loss due to CO increases from about zero to 8%. The stack gas loss increases between the indicated combustion rates from about 7% to 15%, and the losses due to hydrogen and moisture in the coal remain constant at values of about 5% and 2% respectively. The greatest loss at high rates of combustion is that due to sparks and cinders.

Tests made upon a Baltimore & Ohio locomotive at the University of Illinois show in general the same heat losses and verify the results shown in Fig. 2.

The essence of all of Mr. Anthony's articles is contained in Fig. 3, page 24. Curve 1 shows the true furnace efficiency. Furnace efficiency may be defined as the ratio of the heat liberated by the furnace to that contained in the coal fired. The vertical distance above curve 1, then, represents losses due to inefficiency of the furnace. As shown in the figure this loss embraces a large part of the total heat in the coal, especially at high rates of combustion. The vertical distance between curves 1 and 2 at any point represents the percentage of heat that is unavailable for absorption by the boiler. This is an inherent loss, and represents heat which is absolutely unavailable for absorption by the boiler. The vertical distance between curves 2 and 3 represents the percantage of heat that is available for absorption by the boiler, but which is rejected by it. Curve 3, then, represents overall boiler efficiency.

The distance between curves 2 and 3, of Fig. 3, is almost

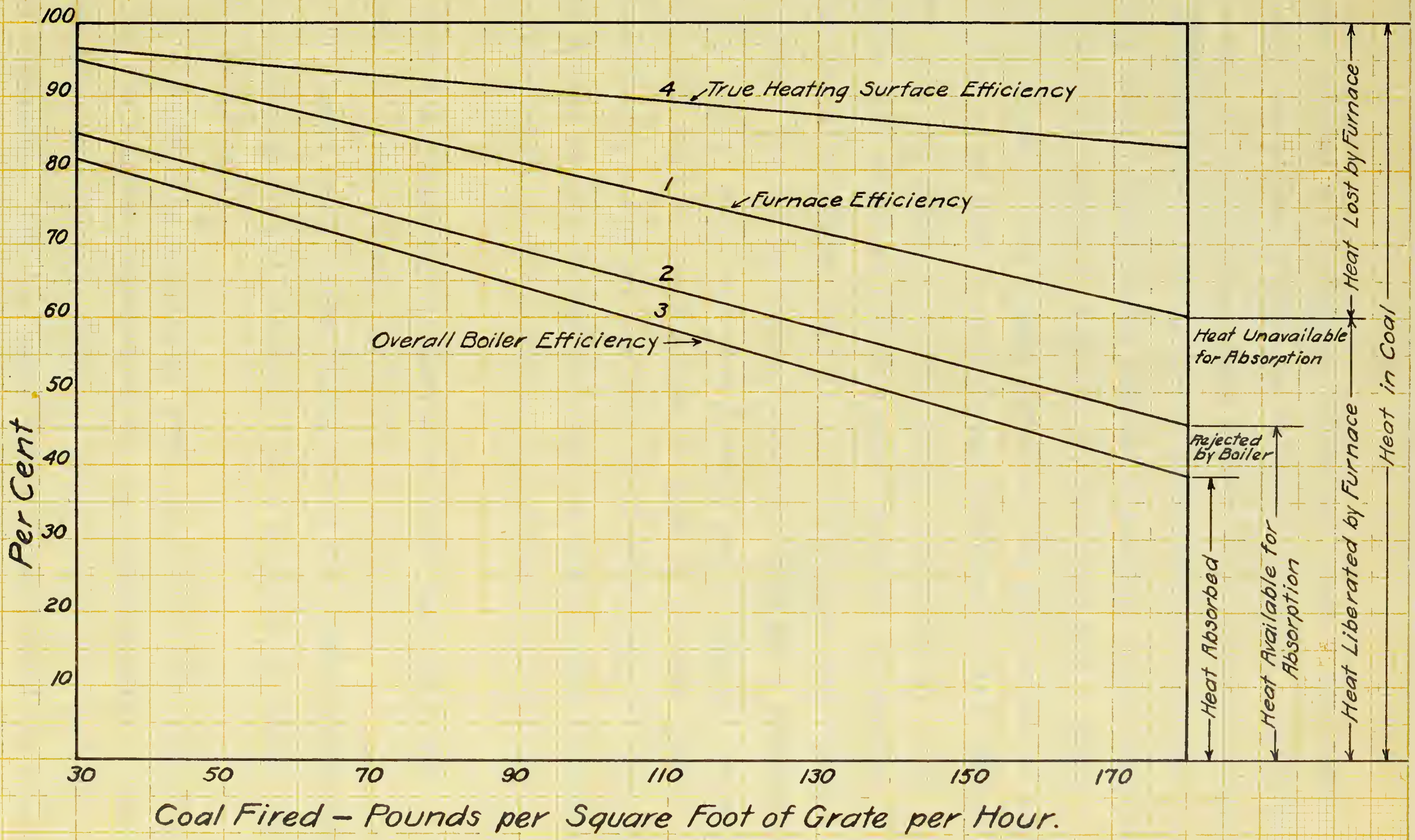

FIG. 3 – EFFECT ON FURNACE AND BOILER EFFICIENCIES OF INCREASING RATE OF COMBUSTION.

constant between the limits of combustion shown. This means that the true boiler efficiency is almost constant, regardless of the rate of combustion. True boiler efficiency may be defined as the ratio of heat absorbed by the boiler to heat available for absorption, and this seems a much fairer method of stating boiler efficiency than the older method of overall boiler efficiency. Since curves 1, 2, and 3 are almost parallel, the drop in the overall boiler efficiency must be due to the inability of the furnace to burn fuel efficiently at high rates of combustion. Fig. 3 indicates that there is a much larger field for increasing firebox efficiency than there is for increasing true boiler efficiency. The boiler heating surfaces, when kept reasonably free from scale, soot, or other foreign deposits, will transmit all the heat that can be imparted to them by the hot gases. The conclusion, therefore, is that the inefficiency of the entire boiler is due to the inability of the firebox to burn fuel efficiently, particularly at high rates of combustion.

Summary.

From the foregoing discussion it would seem that the following important features should be emphasized:

(1) Large grate areas, large firebox volumes, and moderate rates of combustion are necessary for a locomotive firebox to operate efficiently.

(2) In burning bituminous coal, some provision should be made for the complete combustion of the volatile gases. This is accomplished by providing means by which the path of the gases may be lengthened, and the gases kept in the firebox for a greater period of time.

(3) Firebox evaporation depends largely upon the area and temperature of the radiating surfaces. The amount of heating surface in the firebox is not of primary importance, and is incidental to the grate area and firebox volume.

(4) The efficiency and capacity of a boiler does not depend necessarily upon the amount of boiler heating surface used. The heating surface will absorb efficiently the heat that the firebox liberates.

(5) The efficiency of a tube is directly proportional to the length and inversely proportional to the diameter of the tube. Therefore, within certain practical limitations, boiler efficiency may be increased by the use of smaller tubes. The ratio of length to diameter of tubes should be about 100 to 1.

(6) The favorable results which have been brought about through the use of the brick arch have already been stated.

Tests which have been made to determine the effect of the combustion chamber on locomotive boiler performance have brought about the following conclusions:

(1) The effect of the combustion chamber upon the total amount of water evaporated by the boiler is negligable. The effect of the addition of a combustion chamber to most existing locomotives would probably not greatly increase the total amount of water evaporated at comparatively high rates of combustion. A gain would occur at all ordinary rates of combustion through increased firebox efficiency.

(2) By means of the combustion chamber the efficiency of the firebox is increased, due to more complete combustion of the volatile gases and of the small particles of coal which are likely to be carried out through the tubes.

PART IV.

Test Results and their Analysis.

The curves and data shown in this section were compiled to show the effects of the brick arch and combustion chamber upon firebox and boiler performance. Figure 8 shows the effect of varying the grate area while holding the firebox volume constant.

The Brick Arch - Coatesville Tests.

In "Tests of a Jacobs-Shupert Boiler," by Dr. W. F. M. Goss data are presented which show the effect of the brick arch on evaporation performance of a locomotive boiler. Tests were made upon the boiler with and without an arch in the firebox.

The coal used on some of these tests was Scalp-Level, a bituminous coal relatively low in volatile matter but high in fixed carbon. The other tests were fired with Dundon coal, a bituminous coal containing a high percentage of volatile matter and comparatively low in fixed carbon. The table below serves to give a fair idea of the composition of these two grades of coal.

	Moisture, %		Volatile, %		Fixed Carbon, %		Ash, %	
	Min.	Max.	Min.	Max.	Min.	Max.	Min.	Max.
Scalp-Level	1.48	5.70	16.09	18.10	73.02	75.90	4.40	7.07
Dundon	2.27	3.38	32.10	34.34	49.54	52.97	11.84	12.74

Figure 4, page 28, shows that when 6500 lbs. of Scalp-Level coal were being fired per hour the boiler without the arch evaporated 7.35 lbs. of water per pound of coal, while at the same rate of combustion the boiler with the arch evaporated 7.95 lbs. of water per pound of coal. In this case the arch affected a saving of about 8%. When Dundon coal was being fired at the

[illegible]

Comparative Tests of Boilers with and without Brick Arch. - Made in the Series of Coatesville Tests.

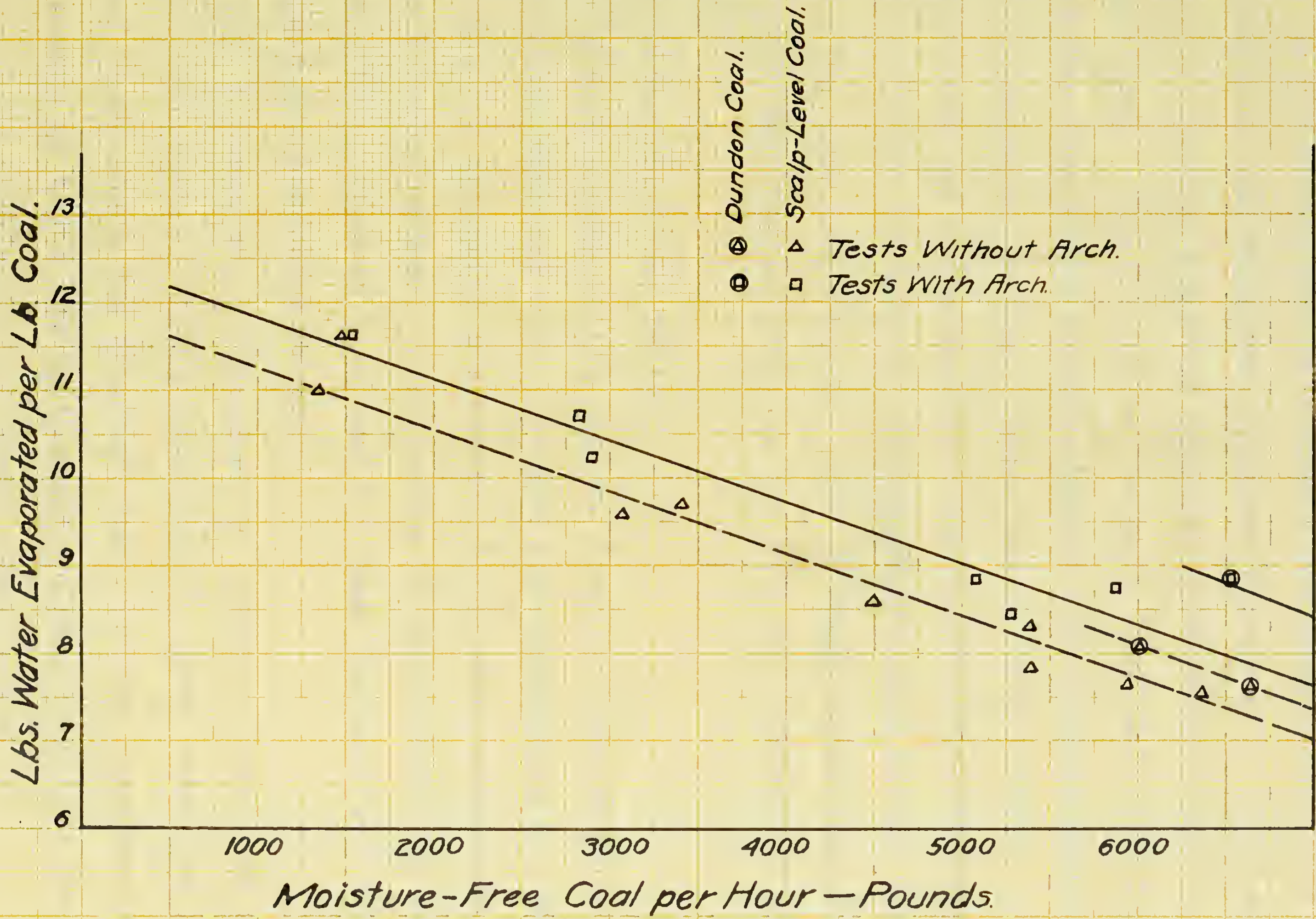

FIG. 4.—EFFECT OF BRICK ARCH ON EVAPORATION.

rate of 6500 lbs. per hour the firebox without the arch evaporated 7.7 lbs. of water per pound of coal, and with the arch an evaporation of 8.7 lbs. of water per pound of coal was obtained. In this case the saving due to the arch was about 12%. Coal with a high volatile content shows a greater increase of efficiency when burned in a firebox equipped equipped with an arch than does low-volatile coal. This is due to the fact that coal with a comparatively low volatile content is burned for the most part upon the grates, because of the high percentage of fixed carbon. A high-volatile coal has a comparatively low fixed carbon content, hence most of its heat is derived from the combustion of the volatile gases. The arch lengthens the path of these gases during combustion, thus insuring an intimate and complete mixture of gases and oxygen.

The Brick Arch--P. R. R. Tests.

Tests were made at the Altoona Testing Plant of the Pennsylvania Railroad upon Atlantic type locomotive No. 5266, class E2A, to determine the effect of the brick arch upon firebox and boiler performance. The results of these tests are shown graphically by Figures 5, 6, and 7. Penn Gas coal was used throughout all tests of this series.

Fig. 5, page 30, illustrates the effect of the arch upon firebox temperature. Tests made with the arch show that a higher firebox temperature was obtained than when the arch was not used. Both series of tests are indicated upon the diagram. Run-of-mine coal was used throughout all tests shown in Fig. 5.

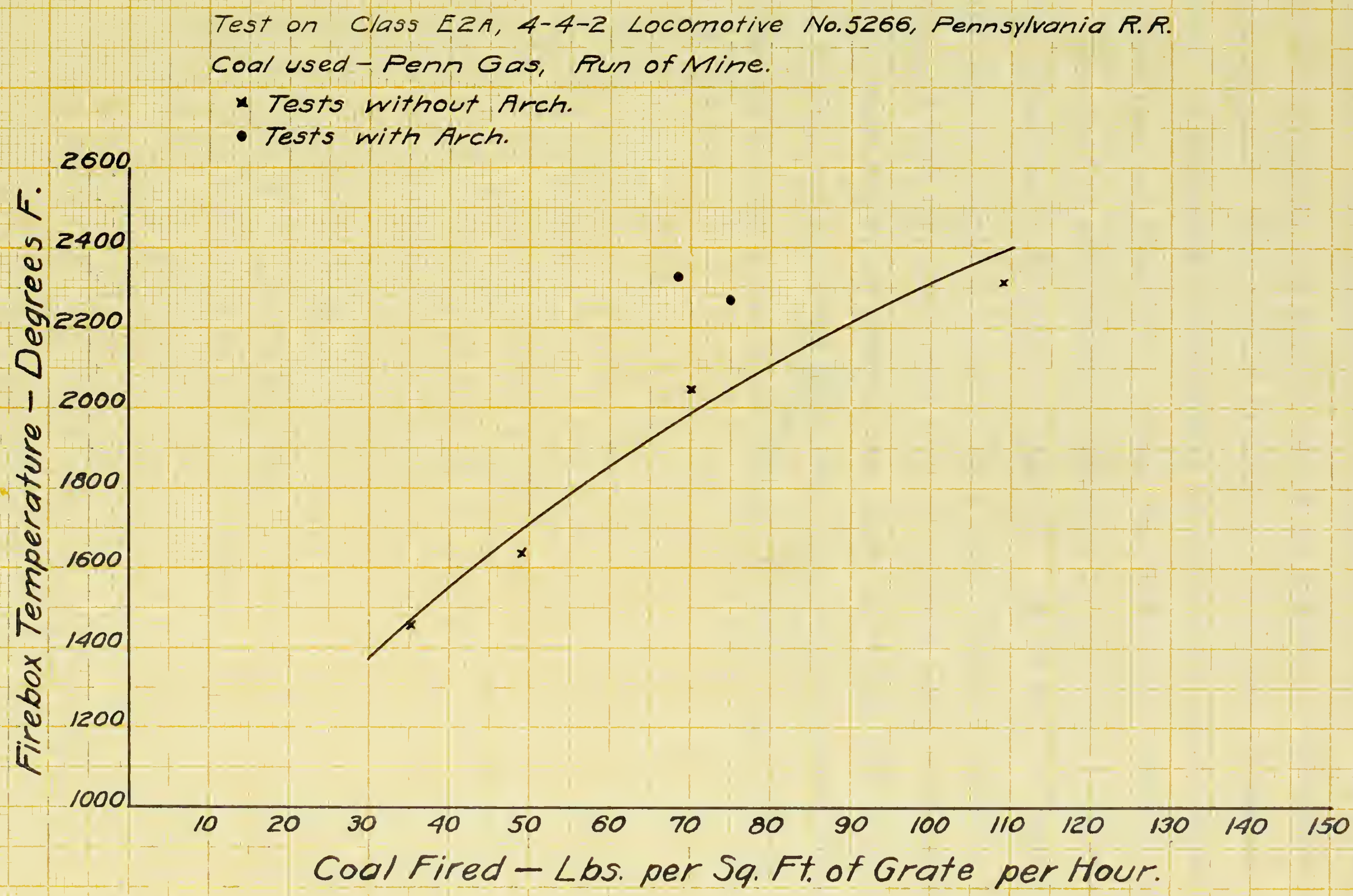

FIG. 5. -EFFECT OF BRICK ARCH ON FIREBOX TEMPERATURE.

Fig. 6, page 32, shows the effect of the brick arch upon evaporation. Tests indicated by points 10 and 11 were made with run-of-mine coal, and points 12 to 16 indicate tests made with screened coal. Tests made without the arch are indicated by the curves. In general, an increase of from 8 to 15% in equivalent evaporation per pound of dry coal was realized by use of the arch.

The boiler efficiencies realized in this series of tests are shown graphically by Fig. 7, page 33. As in Fig. 6, tests indicated by points 10 and 11 were made with run-of-mine coal, and those indicated by points 12 to 16 were made with screened coal. Tests shown by curves are those in which there was no arch in the firebox. In all tests made with the arch an increased boiler efficiency is recorded, this increase varying from about 4% to 14%.

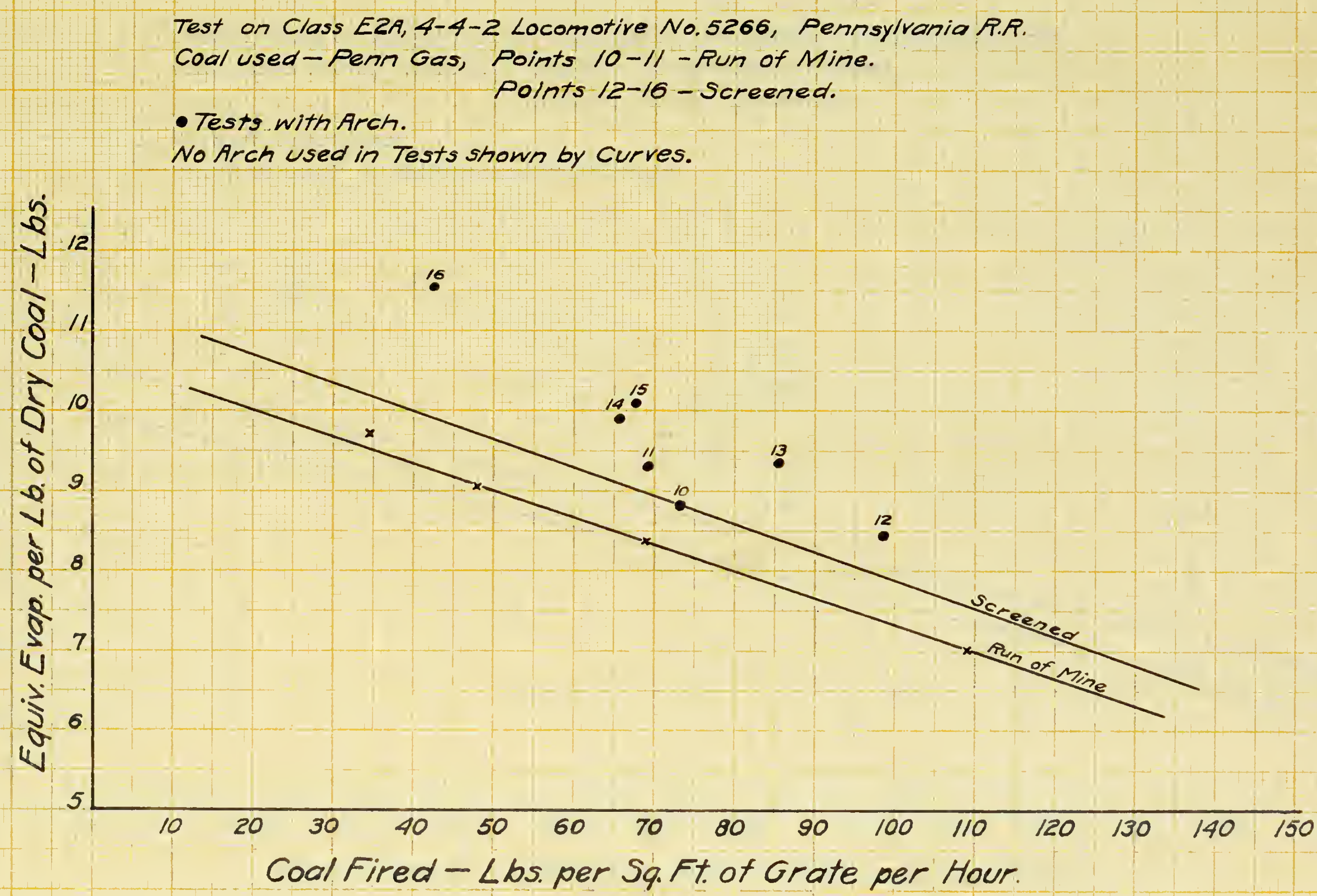

FIG. 6 — EFFECT OF BRICK ARCH ON EVAPORATION.

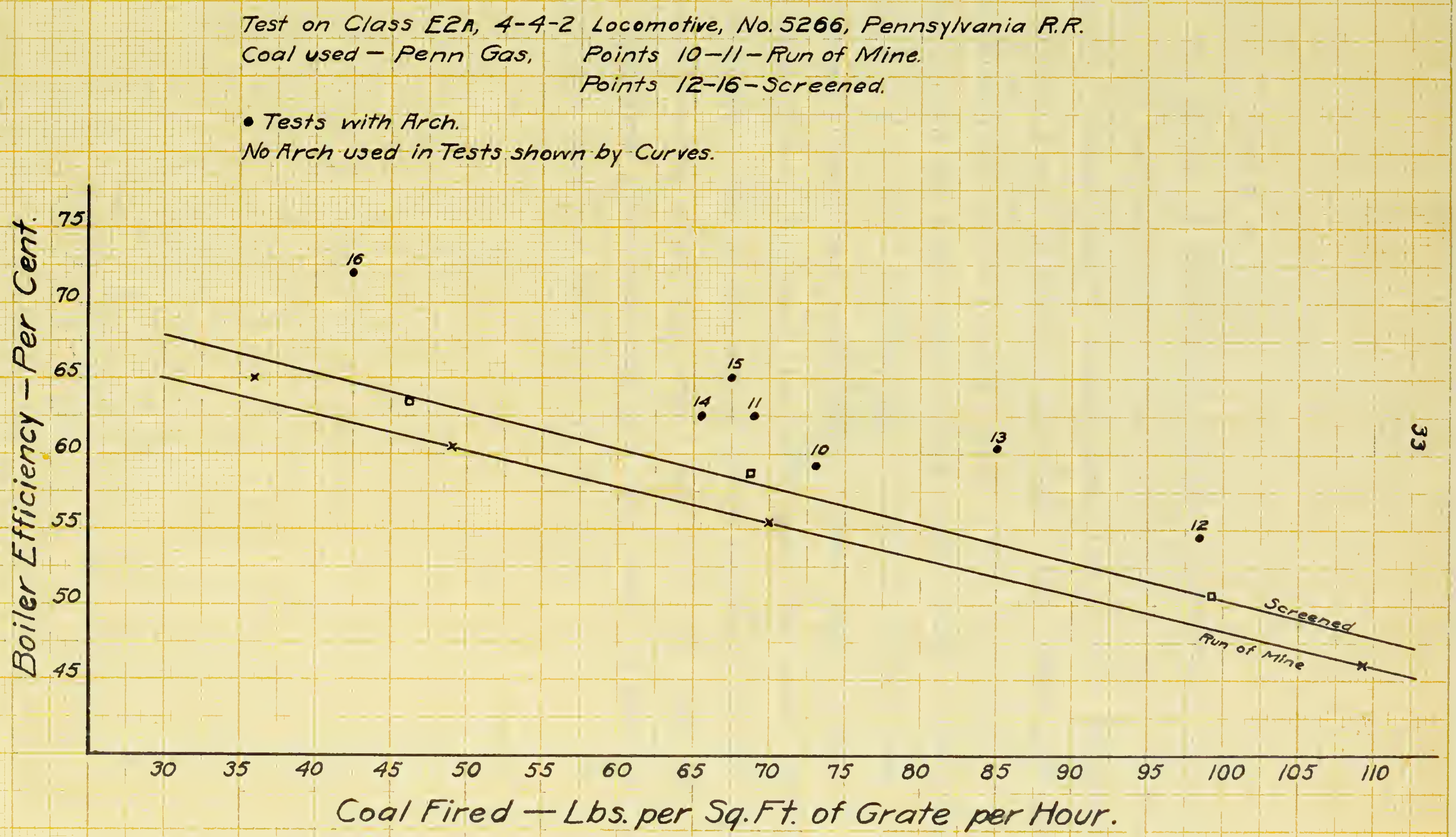

FIG. 7 — EFFECT OF BRICK ARCH ON BOILER EFFICIENCY.

Grate Area, - P. R. R. Tests.

The effect of decreasing the grate area while the firebox volume and heating surface remain constant is shown by Fig. 8, page 35. These tests were made upon Pennsylvania Railroad Class E2A Atlantic type locomotive No. 5266. The normal grate area of this locomotive is 55.5 square feet. Runs were made in which the normal grate was used, and in later tests part of the grate was blocked off so that the area was only 29.76 square feet. The boiler efficiency was considerably higher when the normal grate was used than it was when the small grate was used. The increase of efficiency due to the larger grate decreased as the combustion rate increased. This difference was about 15% when 1500 lbs. of dry coal were fired per hour, but only about 4% when the hourly combustion rate was 6000 lbs. of dry coal.

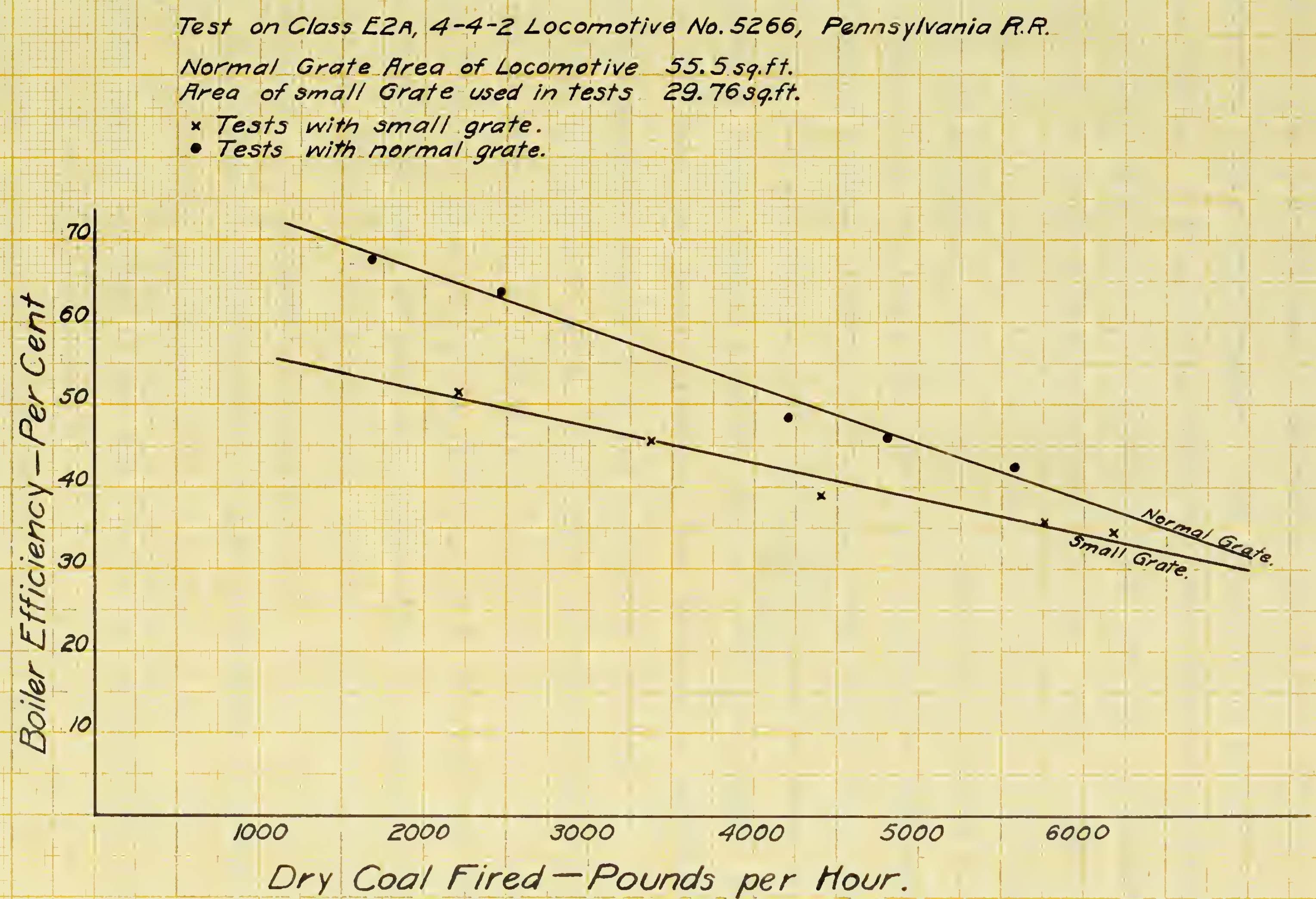

FIG. 8.—CURVES SHOWING EFFECT OF DECREASING GRATE AREA.

Combustion Chamber, - P. R. R. Tests.

Tests were conducted by the Pennsylvania Railroad upon two Pacific type locomotives to determine the effect of the combustion chamber on boiler performance. The locomotives tested were Class K4S No. 1737 and Class K2SA No. 877. These locomotives are almost identical with the exception that the K4S is equipped with a combustion chamber while the K2SA is not. Fig. 9, page 37 illustrates the effect of the combustion chamber on evaporation. The curves show that at low rates of combustion, below 50 lbs. per square foot of grate per hour, the locomotive without the combustion chamber evaporates slightly more water per pound of coal than the locomotive equipped with combustion chamber. At combustion rates higher than 65 lbs. of coal per square foot of grate per hour the locomotive with the combustion chamber evaporates the greater amount of water per pound of coal.

Fig. 10, page 38 shows the effect of the combustion chamber on boiler efficiency. The data illustrated in this diagram were taken from the same tests as illustrated in Fig. 9. At combustion rates below 50 lbs. of coal per square foot of grate per hour the locomotive without the combustion chamber showed the higher boiler efficiency. As combustion increased from 65 to 170 lbs. of coal per square foot of grate per hour the percentage increase in boiler efficiency in favor of the locomotive with the combustion chamber varies between 0 and 5%, the higher efficiency being at the high rate of combustion.

Figure 11, page 40, shows that a boiler which had a combustion chamber evaporated more water and operated at a higher efficiency than one in which there was no combustion chamber. At low rates of

Test on K4s Locomotive No. 1737, 4-6-2 Type, and K2sa Locomotive No. 877, 4-6-2 Type, P.R.R.

Combustion Chamber on No. 1737. No Combustion Chamber on No. 877.

Locomotive No.	1737	877
Class	K4s	K2sa
Cylinders	27×28	24×26
Heating Surface (Tubes)	3728.64	3436.4
Heating Surface (Firebox - incl. Arch Tubes)	306.7	208.
Heating Surface (Superheater)	1171.8	989.
Grate Area	69.26	53.72

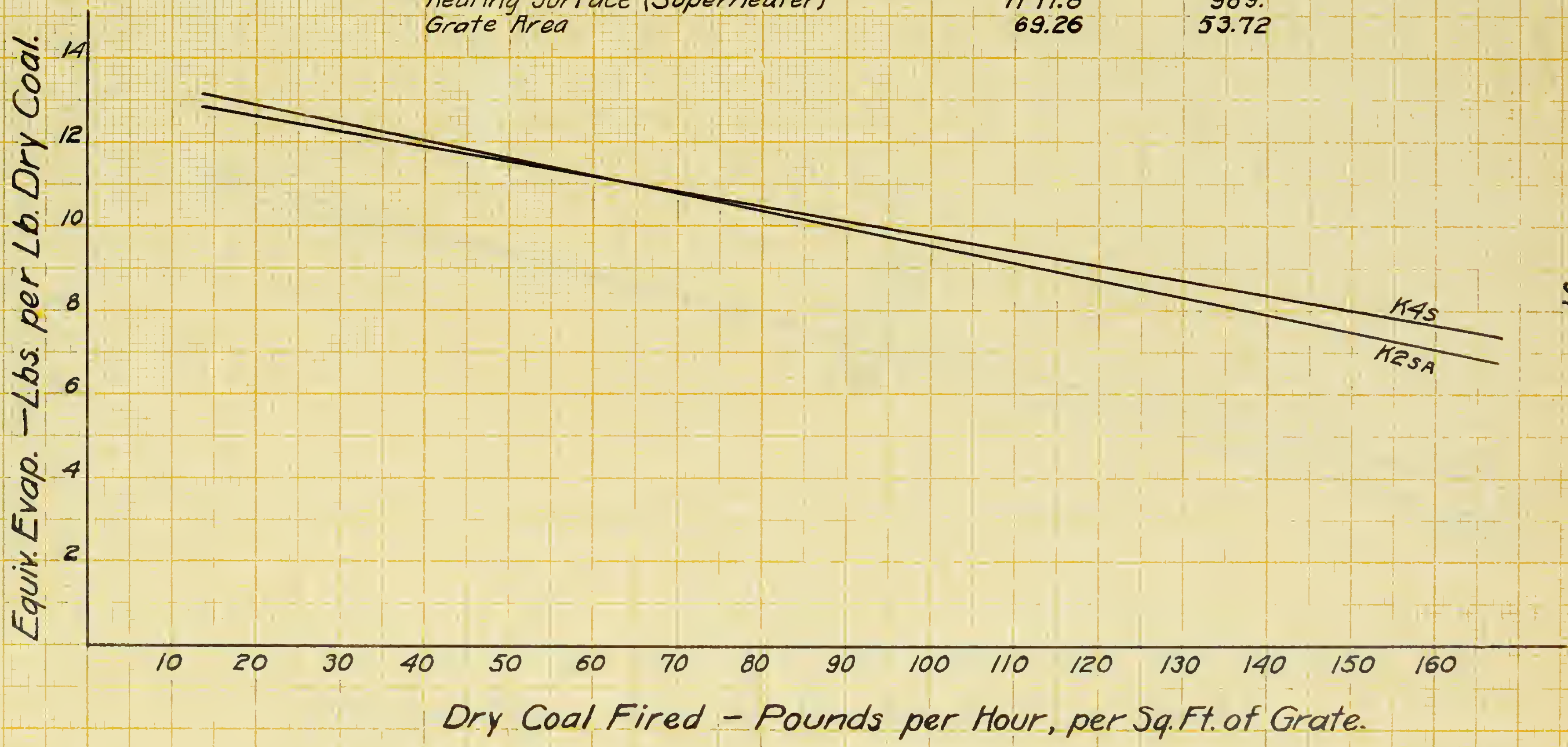

FIG. 9 - CURVES SHOWING EFFECT OF COMBUSTION CHAMBER UPON EVAPORATION.

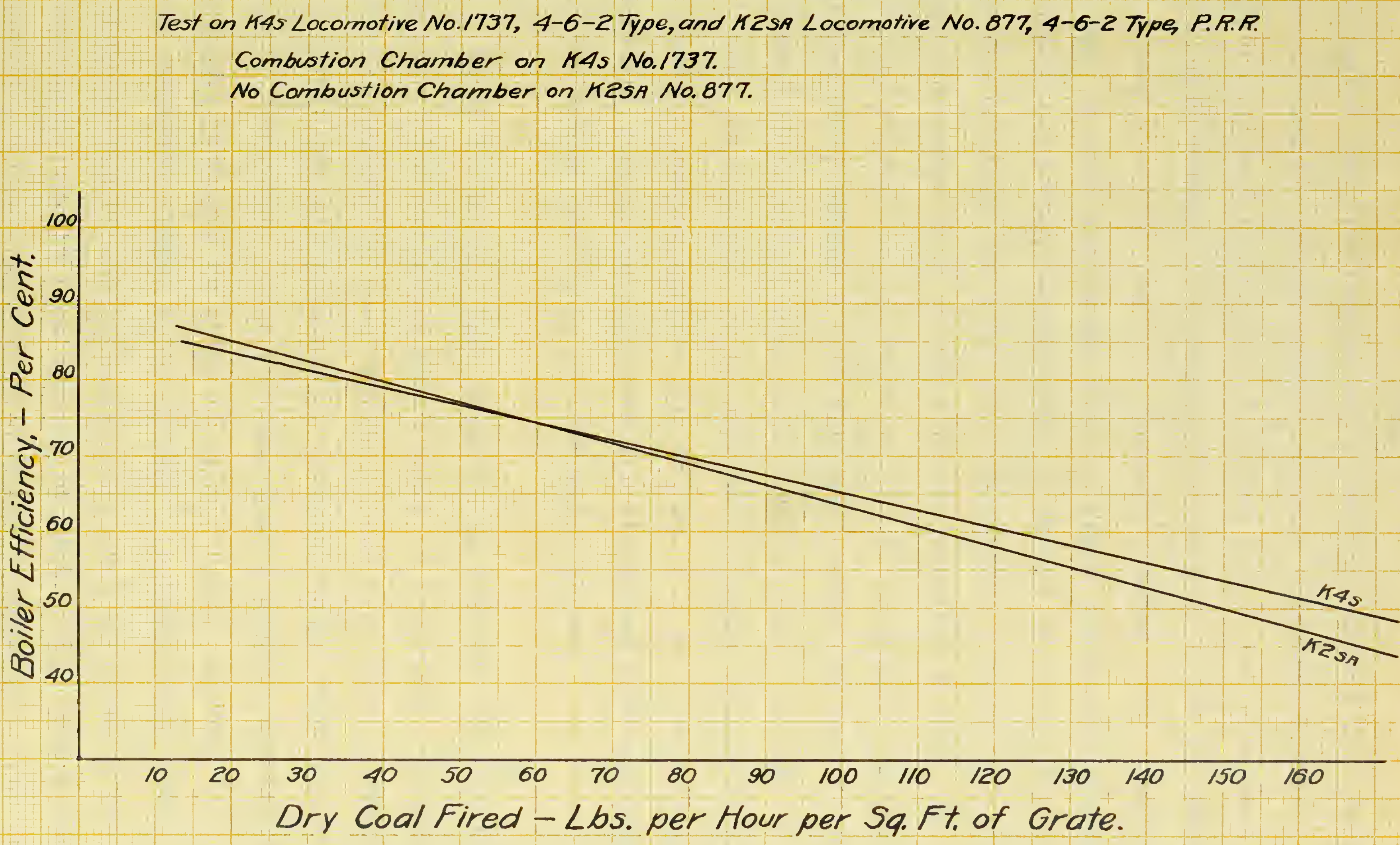

FIG. 10. — CURVES SHOWING EFFECT OF COMBUSTION CHAMBER ON BOILER EFFICIENCY.

combustion the evaporation and efficiency of the two boilers were the same. As the combustion rate increased however, the evaporation and efficiency of Boiler A, the boiler with a combustion chamber, increased with respect to the performance of Boiler B, which had no combustion chamber. For example, at 37 lbs. of coal per square foot of grate per hour, each boiler shows an equivalent evaporation of 32,000 lbs. of water per hour, and an efficiency of 80%. At a combustion rate of 120 lbs. of coal per square foot of grate per hour Boiler A shows an evaporation of 75,000 lbs. of water per hour, as against 65,000 lbs. evaporated by Boiler B. At the same combustion rate boiler A performs at an efficiency of 60%, and boiler B at 50%.

The substitution of a combustion chamber adding 74 sq. ft. of firebox heating surface for 583 sq. ft. of tube heating surface increased very materially both the efficiency and capacity of the boiler through the range of combustion shown.

Summary.

While it has not been possible to consider all of the conditions and variables which might be important in connection with the preceding test results, those results as a whole clearly illustrate certain features of firebox design which are in general desirable. Thus the data just presented emphasize in particular matters relative to brick arches, grates, and combustion chambers. Figs. 4, 5, 6, and 7 indicate that boiler efficiency can be substantially increased through the use of the brick arch, roughly from 5 to 10 percent. Fig. 8 indicates that increasing grate area may be of value in raising boiler efficiency, also to the extent of 5 to 10 percent. Figs. 9, 10, and 11

	Grate Area	F.B.H.S.	F.B.Vol.	Tube and Flue H.S.	S.H.S.	Total H.S.
Boiler A.	70	306	427	3385	1172	4863
Boiler B.	70	232	311	3968	1302	5502

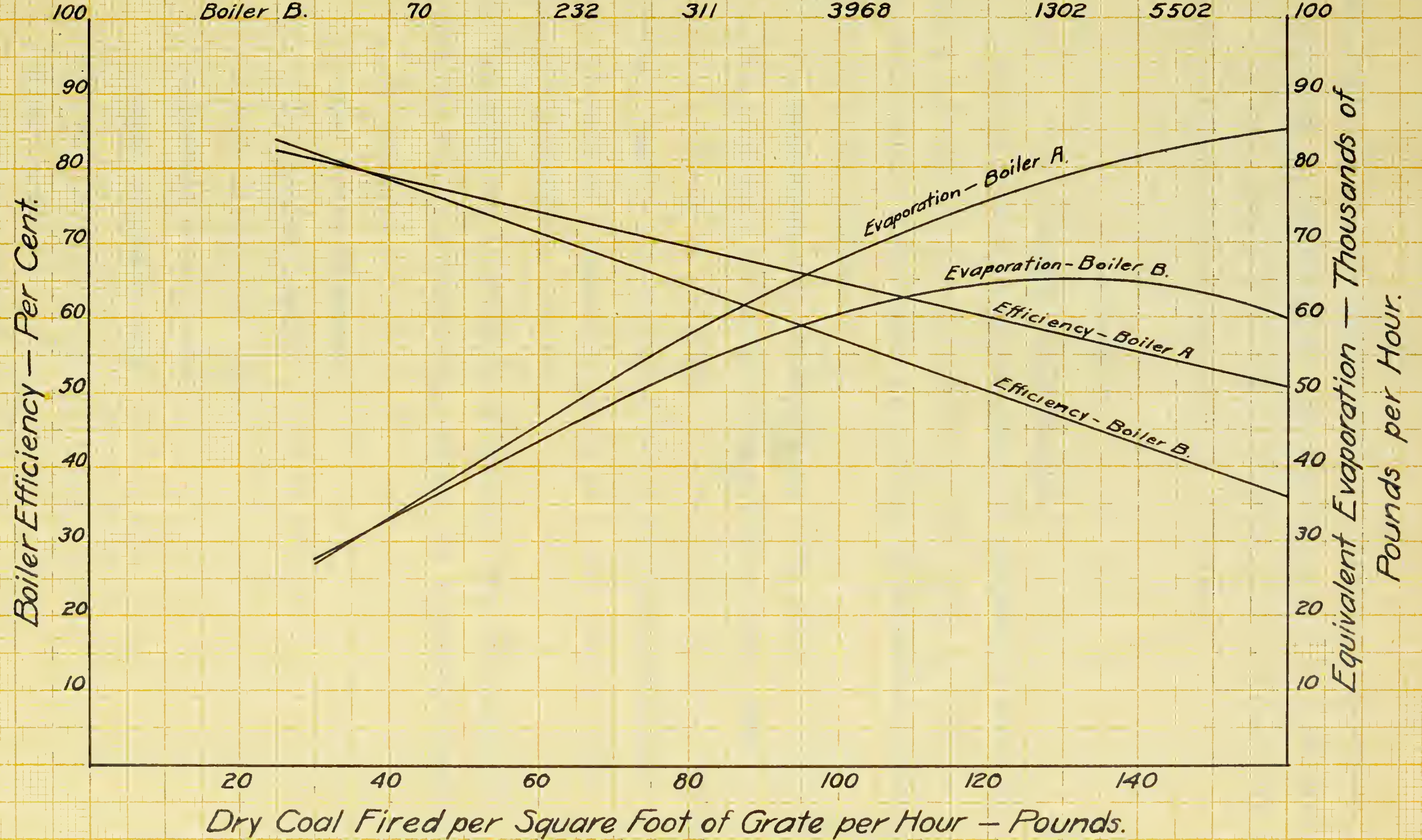

FIG. 11.— CURVES SHOWING EFFECT OF COMBUSTION CHAMBER ON BOILER PERFORMANCE AND EFFICIENCY.

indicate that the use of the combustion chamber may increase boiler efficiency, possibly to the extent of 10 or 15 percent. It will be noted that all of these improvements increase boiler efficiency through increasing furnace efficiency.

PART V.

General Conclusions.

1. There is greater opportunity to increase overall boiler efficiency through improved furnace design, thus increasing furnace efficiency, than there is through improvements relating solely to heating surface.

2. Further study and experimental work should be carried on in connection with combustion processes as they occur in locomotive fireboxes to determine the relation of those processes to different features of design under different conditions relative to rate of combustion, kind of fuel, size of coal, amount and character of ash, etc.

3. Improvement in furnace efficiency may be looked for in connection with the following matters or locomotive parts:

Air supply.

(a) Grate openings.

(b) Ashpan design.

Proper mixture of gases.

Temperature in firebox.

Length of flame path.

Firebox volume.

Grate area.

Radiating surfaces.

Tube heating surface.

(a) Velocity of gases.

Firebox heating surface.

Firebrick arch.

Combustion chamber.

Kind of fuel and fuel characteristics.

Careful consideration should be given to these matters when designing new equipment and when existing equipment is to be overhauled or remodeled.

4. It therefore seems suitable to conclude that the greatest advantages in economy will be gained,

1st. by giving careful attention to improving combustion processes. This will be possible largely through a thorough understanding of the principles of combustion, particularly as they apply to the locomotive firebox.

2nd. by increasing firebox volume. This may be accomplished through the use of the combustion chamber and by increasing grate area.

3rd. by increasing firebox temperature and length of flame path, thus producing more complete combustion. This may be accomplished through the use of the firebox arch.

Acknowledgements.

In compiling information and facts concerning questions relative to the subject, the writers have consulted various standard texts, railway journals, and test data. In this connection they acknowledge the use of information from the following sources:

"Steam Boiler Economy," - Kent.

"Heat Transmission Through Boiler Tubes," - Kreisinger and Barkley, - U. S. Bureau of Mines Technical Paper No. 114.

Report of Committee on Fuel Tests, International Railway Fuel Association, 1917.

"Tests of a Jacobs-Shupert Boiler," - Goss.

"Firebox Volume and Boiler Efficiency," - Anthony, in "The Railway Age Gazette," April 20, 1917.

"Locomotive Boiler Efficiency," - Anthony, "Railway Mechanical Engineer," Volume 90, p. 443.

"Locomotive Boiler Efficiency," - Anthony, "Railway Mechanical Engineer," Volume 90, p. 505.

Bulletin No. 8, Pennsylvania R. R., "Tests of an E2A Locomotive."

Bulletin No. 29, Pennsylvania R. R., "Tests of a K4S Locomotive."